

Low temperature Stirling engines pressurised with real gas effects



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ABSTRACT

With the aim of improving the performance of Stirling engines for applications with relatively low maximum temperatures (150 °C–300 °C), working fluids that have noticeable real gas effects have been considered.

The simulation model developed for the preliminary analysis of Stirling engines takes into account the losses of mechanical power and the efficiency of the regenerator with two adimensional coefficients, on the basis of experimental data available in literature. Comparisons between the working fluids were made having fixed the value of these two parameters. The results from this simplified analysis can be taken into consideration when evaluating the potential of these Stirling engines with real gas. They revealed that an engine cycle with a maximum pressure 3–4 times greater than the critical pressure and using a fluid with a critical temperature close to the minimum temperature of the engine cycle, gives an increase in power about 2.5 times greater than the case with ideal gas and an increase in cycle efficiency of around 50 percent. Such a result means that the use of fluid mixtures as the working fluids now becomes interesting, since the critical point of mixtures can be gradually adapted to meet design requirements.

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1. Introduction

To date, it is still fossil fuels that are most commonly used in the production of thermal energy and electricity. Nevertheless, recent decades have seen a growing interest in high efficiency energy conversion systems that aim to make better use of renewable energy resources. Within this context, the Stirling cycle offers interesting possibilities.

These engines have been proposed for CHP (combined heat and power) [1] and micro-CHP applications [2], for small-scale energy production from a few hundred watts [3] up to several kW [4] and for exploiting low-temperature heat sources [5] (from 110 °C up to 200 °C). Furthermore, Stirling engines are often considered as alternatives to internal combustion engines in the automotive sector [6].

Traditionally, Stirling engines have been pressurised by fluids that, under normal working conditions, have a similar behaviour to ideal gas. The most commonly used working fluids are air, nitrogen, helium and hydrogen. However, engines that are pressurised with these working fluids have two great drawbacks. The most

important of these, a low specific power compared to internal combustion engines.

For example, the engine described in Ref. [7] has a power of about 35 kW at 1040 rpm with a work volume of 4814.38 cm³ and a maximum cycle temperature of around 700 °C. Another example is the Ford-Philips 4-215 engine (built at the end of the 1980s, with a total internal volume of 3500 cm³) [8,9], which has a useful output of 200 kW at 4500 rpm at the maximum temperature of 750 °C, with hydrogen as the working fluid and an average pressure of 200 bar.

Secondly, in order to get a good performance from these engines, the working fluid needs to be raised to very high pressure and temperature levels. Such working conditions means using materials that are highly resistant to stress and very costly.

So far, to improve the Stirling engines performance the research was focused on the development of the mechanical transmission system [10] and on the gas circuit performance [11]. In particular looking for driver mechanism to minimise mechanical losses and heat exchanger geometry to reduce pressure losses and improve heat exchange. The choice of working fluids is at present limited to helium, hydrogen and air (nitrogen).

This article analyses the effects caused by the use of working fluids under thermodynamic conditions, which, compared to the usual behaviour of an ideal gas, give rise to significant increases in useful power and in engine efficiency.

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Nomenclature

Symbols

Be	Beale number, $(= \dot{W}/P_{\max} V_{\text{swf}})$
c_p, c_v	mass heat capacity, kJ/kgK
f	frequency, Hz
h	heat transfer coefficient, W/m ² K
M_M	molar mass, g/mol
M	mass, kg
$\dot{m}$	mass flow, kg/s
n	revolution per minute, rpm
P	pressure, bar
Pr	Prandtl number
Q	thermal energy, J/cycle
$\dot{Q}$	thermal power, kW
R	gas constant, J/kgK
Re	Reynolds number
s	specific entropy, J/kgK
S	entropy, J/K
S_{irr}	entropy produced by irreversible processes
t	time, s
T	temperature, °C
v	flux velocity, m/s
V	volume, m ³
V_{cl}	total clearance volumes, m ³
$V_{\text{clc}}, V_{\text{cle}}$	compression and expansion clearance volumes, m ³
V_{sw}	total swept volumes, m ³
$V_{\text{swc}}, V_{\text{swe}}$	compression and expansion swept volumes, m ³
W	engine work, J/cycle
$\dot{W}$	mechanical power, kW
α_p	isobaric coefficient of thermal expansion
ε	regenerator efficiency parameter ($\min(\varepsilon_K, \varepsilon_H)$)
ϕ	phase angle
η	fluid dynamic coefficient
η_{cy}	cycle efficiency

η_{Carnot}	Carnot cycle efficiency $(=1-T_{\min}/T_{\max})$
κ	thermal conductivity, W/mK
κ_T	isothermal compressibility coefficient, W/mK
ρ	density, kg/m ³
ω	acentric factor
ω_s	angular velocity, rad/s $(=2\pi N/60)$
μ	dynamic viscosity, Pa·s
ρ	density, kg/m ³
τ	temperature ratio, $(=T_{\max}/T_{\min})$
$\dot{X}$	time derivative of X , $(=dX/dt)$

Subscripts

cr	critical point of the fluid
max	maximum
min	minimum
r	reduced condition $(T_r = T/T_{\text{cr}}, P_r = P/P_{\text{cr}})$
sat	saturation point of the fluid
C	compressor
E	expansor
R	regenerator
H	heater
K	cooler
T	total
D	dead
CK	physical quantity on the interface between Compressor and Cooler
KR	physical quantity on the interface between Cooler and Regenerator
RH	physical quantity on the interface between Regenerator and Heater
HE	physical quantity on the interface between Heater and Expansor

Acronyms

CHP	combined heat and power
EOS	equation of state

J.F. Malone in Ref. [12] was the first to propose using working fluids under conditions that differed from those of an ideal gas in Stirling engines. Malone designed an engine pressurised by water under super-critical conditions. Subsequently, his ideas were studied by other authors [13–15].

The purpose of this work is to improve and expand upon the analysis proposed in Ref. [15], optimising the corrective coefficients of the numeric model on the basis of experimental data drawn from literature and extend the thermodynamic analysis to different working fluids, both pure and mixed.

In Ref. [15] it was also shown that an engine cycle with real gas effects can increase both power and efficiency of the Stirling engine. Mixing two substances of different critical temperatures yields an infinite number of fluids with different critical point [16] and enables the exploitation of real gas effects in all operating condition. For this reason their study, in Stirling engine application, has particular interest.

2. The engine model

We took the so-called Adiabatic Model as our model [8]. The model was then modified by applying two corrective coefficients, which were introduced to account for the typical power and heat losses of these engines [15].

In the calculation model, created by using the commercial program aspenONE®, the engine is subdivided into five volumes: the compression volume (Compressor), the cooling volume (Cooler), the regenerative volume (Regenerator), the heating volume (Heater) and the expansion volume (Expander). As shown in Fig. 1 the heat exchangers are included between the two pistons.

The model is based upon the following hypotheses:

- the compression, the expansion and the regenerator volumes are adiabatic

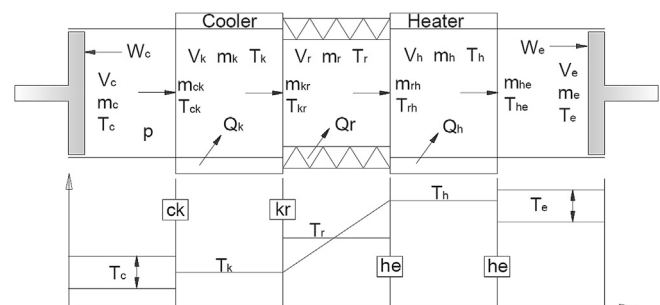


Fig. 1. Schematic Adiabatic Model [8].

- each volume is assumed to have a uniform temperature distribution. In particular, the temperature of the Cooler, Heater and Regenerator are constant and the temperature of the Regenerator is the logarithmic mean between the temperature at the Cooler–Regenerator interface and the Regenerator–Heater
- no leakage is present
- the pressure is uniform inside the engine

The volumes of the Compressor and the Expander follow a simple sinusoidal law.

with, $\omega_s = 2\pi N/60$ angular speed.

To resolve the cycle and calculate the engine performance on the basis of our calculation model, we applied the laws of mass and energy conservation to each control volume. To describe the volumetric properties of the working fluid, the equation of state used in the model (both for pure fluids and mixtures) is the Penge–Robinson [17] (Equation (1)), modified by G. Soave with the introduction of the function $\alpha(T)$ [18] (Equation (2)).

$$P = \frac{RT}{v-b} - \frac{a}{v(v+b) + b(v-b)} \quad (1)$$

in which

$$\begin{aligned} b &= \sum_{i=1}^n x_i b_i \\ a &= \sum_{i=1}^n \sum_{j=1}^n x_i x_j \sqrt{a_i a_j} (1 - k_{ij}) \\ b_i &= 0.07780 \frac{RT_{c,i}}{P_{c,i}} \\ a_i &= \alpha_i(T) 0.45724 \frac{R^2 T_{c,i}^2}{P_{c,i}} \\ \alpha_i(T) &= \left[1 + m_i \left(1 - \sqrt{T_{r,i}} \right) \right]^2 \\ m_i &= 0.37464 + 1.54226 \omega_i - 0.26992 \omega_i^2 \end{aligned} \quad (2)$$

with x_i indicating the molar fraction of the component i —*esimo*, k_{ij} with the coefficients of binary interaction of the mixtures and ω_i is the acentric factor of the component i —*esimo*.

The coefficients of binary interaction k_{ij} , which, for some of the mixtures considered, were not found in the database of aspenONE®, have been optimised on the basis of experimental data drawn from literature.

The design parameters are:

- the Compressor and the Expander swept volumes (V_{swc} and V_{swe})
- the dead engine volume, V_D (V_K , V_R , V_H , V_{clc} and V_{cle})
- the Cooler (T_K) and the Heater temperatures (T_H)
- the phase angle between the Compressor and the Expander volume variation

The physical quantities to be calculated are.

- the temperatures of the working fluid in the Compressor and in the Expander ($T_C(t)$ and $T_E(t)$)
- the engine pressure ($P(t)$)
- the mass contained in each control volume ($M_C(t)$, $M_K(t)$, $M_R(t)$, $M_H(t)$, $M_E(t)$)
- the temperature at the interface between the Cooler and the Regenerator (T_{KR}) and at the interface between the Regenerator and the Heater (T_{RH})

Our calculation model had two coefficients introduced.

The first, ε , defines the efficiency of the Regenerator. The Regenerator is a heat accumulator. In general, it consists of a thick mesh or channels, through which the gas, in cycles, releases and accumulates heat in the matrix.

The Regenerator is the most important element in guaranteeing the engine efficiency; if it is designed properly, it recovers most of the perceptible heat of the gas. The coefficient ε is expressed as a ratio between the heat recovered by the Regenerator and the heat ideally recovered if the Regenerator had an infinite surface.

Under these conditions, the temperature difference on the interface of the Regenerator between the Regenerator and the successive control volume ($T_{KR}-T_K$) or (T_H-T_{RH}) is nil. However, since the Regenerator has a finite dimension, there is a temperature difference between the outflow from the Regenerator and the temperature of the adjoining volumes. The parameter ε is defined as follows.

$$\varepsilon = \max(\varepsilon_K, \varepsilon_H) \quad (3)$$

in which

$$\varepsilon_K = 1 - \frac{T_{KR} - T_K}{T_H - T_K} \quad \text{and} \quad \varepsilon_H = 1 - \frac{T_H - T_{RH}}{T_H - T_K} \quad (4)$$

The second parameter, indicated by η , takes into consideration the effects of the power loss which occurs during the engine cycle. It increases the power required in the fluid compression phase and reduces the extractable power during the expansion phases.

The solution is obtained by integrating all the equations until a stable condition is reached. The control variable for evaluating the stability is the heat energy exchanged by the Regenerator over the cycle, which, globally, should be zero ($\oint \dot{Q}_r dt$) [8].

As a further check of thermodynamic consistency the calculations also include an entropic analysis of the various engine components.

The entropy balance equations can be applied equally to all the control volumes (Equation (5)) [19].

$$\frac{dS}{dt} = \sum_i \frac{\dot{Q}_i}{T_i} + \sum_k s_k \dot{m}_k - \sum_j s_j \dot{m}_j + \dot{S}_{irr} \quad (5)$$

where the term on the left indicates the variation over time of the entropy of the control volume in question. The terms on the right of the equals sign indicate, respectively:

- $\dot{Q}_i/T_i$: the entropy exchanged per unit of time per heat type interaction at temperature T_i with $\dot{Q}_i < 0$ if out flowing from the engine;
- $s_k \dot{m}_k$: the entropy exchanged per unit of time per mass flow type interaction (flowing into the system);
- $s_j \dot{m}_j$: the entropy exchanged per unit of time per mass flow type interaction (flowing out of the system);
- $\dot{S}_{irr}$: the entropy produced as a result of the various losses in the control volume per unit of time.

The entropic balance of any component of a Stirling engine is defined by Equation (6). In this equation, i indicates the i -th component of the balance, j is the component that precedes it, whilst k is the component that follows it.

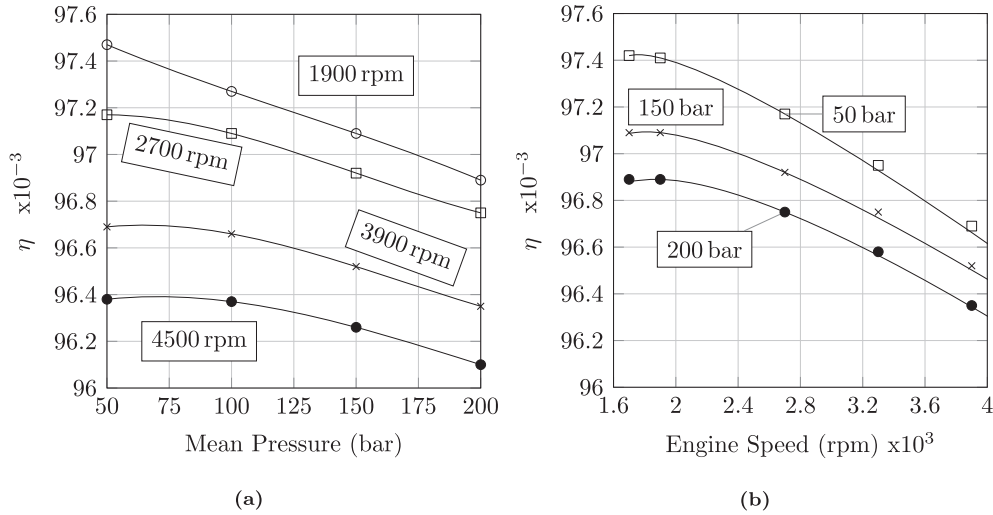


Fig. 2. Ford 4-215 optimisation of fluid dynamic coefficient η ($\varepsilon = 1$). (a) for fixed engine speed and (b) for fixed mean pressure.

$$\dot{S}_{\text{irr}} = -\dot{m}_{ji}s_{ji} + \dot{m}_{ik}s_{ik} - \frac{\dot{Q}_i}{T_i} + \frac{dS_i}{dt} \quad (6)$$

with $s_{ji} = s_i$ if the mass flow goes from i to j or $s_{ji} = s_j$ if the mass flow goes from j to i .

The instantaneous entropic balance is then integrated over the period (Δt) of the engine cycle (Equation (7)).

$$\int_0^{\Delta t} \dot{S}_{\text{irr}} dt = \int_0^{\Delta t} (-\dot{m}_{ji}s_{ji} + \dot{m}_{ik}s_{ik}) dt - \int_0^{\Delta t} \frac{\dot{Q}_i}{T_i} dt + \int_0^{\Delta t} \frac{dS_i}{dt} dt \quad (7)$$

Once the entropy generated by thermodynamic irreversibility has been calculated for each component i of the engine, it is possible to identify the effect this has upon the cycle performance, according to the Equation (8).

$$\Delta\eta_i = \frac{S_{i,\text{irr}} T_K}{Q_H} \quad (8)$$

in which T_K is the temperature of the heat sink and Q_H is the heat exchanged with the heat source.

Adding up the entropic contribution of each component, we can calculate the first law performance of the engine cycle according to Equation (9), in which N indicates the number of components in the system.

$$\eta = \eta_{\text{Carnot}} - \sum_{i=1}^N \Delta\eta_i \quad (9)$$

2.1. The computational model validation and the optimisation of the loss coefficients

To validate the calculation model, results obtained under ideal conditions ($\eta = 1$ and $\varepsilon = 1$) were compared with those reported in Ref. [8], for ideal gas, and the results reported in Ref. [15], calculated with an adiabatic model for real gas.

Under the same working conditions, there was a substantial correspondence between the results and those taken as reference.

In Ref. [9] experimental data are reported for various Stirling engines.

These engines, designed with a variety of mechanisms (rhombic guide of the GPU-3, with the swashplate-drive of the Ford-Philips

4-215) also have a power range, from just a few watts of the Ross Yoke drive, up to several hundred kW of the Ford-Philips 4-215 [8].

The engines were tested under various pressure and temperature conditions and with two different working fluids: H_2 e He .

As a result, at least for these working fluids, it was possible to find a sufficient amount of experimental data to enable an estimate for the coefficients η and ε .

The first evaluation was made with reference to the Ford-Philips 4-215 engine. Experimental data is available for various operating conditions, with varying mean fluid pressure and rotation speed.

The optimisation results are reported in Fig. 2. As can be seen in Fig. 2, η is inversely proportional to both the mean engine pressure and the rotation speed. In particular, it should be noted how, with the engine speed fixed, a variation from 50 bar to 200 bar in the mean pressure causes a reduction of 0.5 percent compared to the reference.

Increasing the engine speed from 1600 rpm to 4400 rpm, fixing the mean pressure of the fluid, leads to a reduction of about 1 percent in the parameter η .

In Fig. 3 the available experimental data are compared with results obtained from our calculation, considering a mean value of η for each rotation speed.

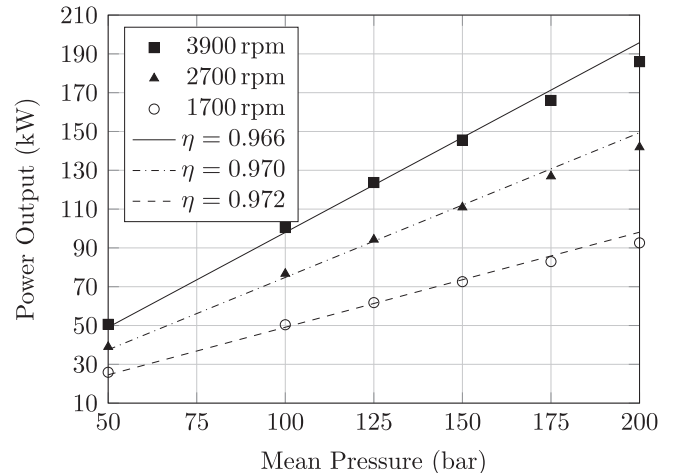


Fig. 3. Ford 4-215 comparison between our model ($\varepsilon = 1$) and Ford-Philips 4-215 experimental data [8].

As far as the Ford-Philips 4-215 engine is concerned, data were only collected for the useful power. To evaluate ε , it is necessary to have data regarding the engine efficiency too.

Some useful data can be found in Ref. [9], regarding the GPU-3, an engine that has been tested with both helium and hydrogen. The optimisation of η is reported in Fig. 4a. It shows that η varies in proportion to the rotation speed. However, the variation of η does not perform in the same way for both helium and hydrogen. The reduction of η is lower for the hydrogen engine than the helium one. The engine that is pressurised with H_2 feels less the effect of the reduction in useful power, caused by the increase in pressure losses, which are proportional to the rotation speed.

The second parameter ε was optimised with regard to the engine efficiency.

In Fig. 4b the values of ε are reported. The figure shows that ε increases proportionally with the engine speed. This is because the convective heat exchange coefficient for the gas flow increases with the speed of the flow itself. The efficiency of the regenerator is heavily influenced by the characteristics of the working fluid. The efficiency is greater in the case of hydrogen.

On the other hand, though, the gas passing through the regenerator at high speed, is subject to high pressure drop and of useful power [20,21]. As a result, once the dimensions of the heat exchangers have been set, it is well known that the choice of working fluid has a significant impact on the overall performance of the engine.

Various authors have calculated the performance of the Regenerator by means of the ratio between the heat recovered per surface unit and per unit of temperature difference (h), and the power to overcome the pressure drop per unit of area ($\dot{W}_f$). As shown in Ref. [22] h and $\dot{W}_f$ are described by Equation (10) and Equation (11).

$$h = \frac{\mu c_p}{Pr^{2/3}} \frac{1}{4r_h} f_h Re = \rho v c_p^{1/3} \kappa^{2/3} \mu^{-2/3} f_h \quad (10)$$

$$\dot{W}_f = \frac{1}{2} \frac{\mu^3}{\rho^2} \left(\frac{1}{4r_h} \right)^3 f_r Re^3 = \frac{1}{2} f_r \rho v^3 \quad (11)$$

whereby, Re is the Reynolds number ($(4r_h \rho v)/\mu$), r_h is the hydraulic diameter, Pr is the Prandtl number ($\mu c_p/\kappa$), κ is the thermal conductivity, $f_h(Re)$ and $f_r(Re)$ are functions dependent upon the type of exchanger used. The ratio between these two parameters is

Table 1

Parameters for the calculation of performance of the thermodynamics cycles.

Design parameter	Symbol	Value
Compression clearance volume	V_{clc}	25 cm ³
Expansion clearance volume	V_{cle}	25 cm ³
Compression swept volume	V_{swc}	50 cm ³
Expansion swept volume	V_{swe}	50 cm ³
Cooler volume	V_k	15 cm ³
Regenerator volume	V_r	75 cm ³
Heater volume	V_h	60 cm ³
Phase angle	ϕ	$\pi/2$
Temperature ratio	$\tau = T_k/T_h$	0.5
Engine speed	n	1500 rpm
fluid dynamic coefficient	η	1–0.98
Regenerator efficiency parameter	ε	1–0.95

$$\frac{h}{\dot{W}_f} = 2c_p^{1/3} \kappa^{2/3} \mu^{-2/3} f_h v^{-2} f_r^{-1} \quad (12)$$

Once the rotation speed is fixed, comparing the two different fluids, we get Equation (13).

$$\left(\frac{h/\dot{W}_f}{h/\dot{W}_f} \right)_1 = \left(\frac{\kappa_1}{\kappa_2} \right)^{2/3} \left(\frac{c_{p1}}{c_{p2}} \right)^{1/3} \left(\frac{\mu_1}{\mu_2} \right)^{-2/3} \quad (13)$$

It is known how the parameters of the Equation (13) depend on the thermodynamic state of the working fluid and, consequently, the choice of the optimal working fluid must be made as a function of the working conditions of the engine.

3. The performances of Stirling engines with real gas effects

The reference design parameters for comparing the performances of engines pressurised by different gases are to be found in Table 1. A list of the fluids considered is in Table 2.

The calculations for each working fluid have been made using the same reduced conditions of temperature and pressure ($T_r = T/T_{cr}$, $P_r = P/P_{cr}$).

For the binary mixtures considered, the interaction coefficients k_{ij} are available for the mixtures: CO_2-CH_4 , $CO_2-C_2H_6$ and for $CO_2-C_7H_8$.

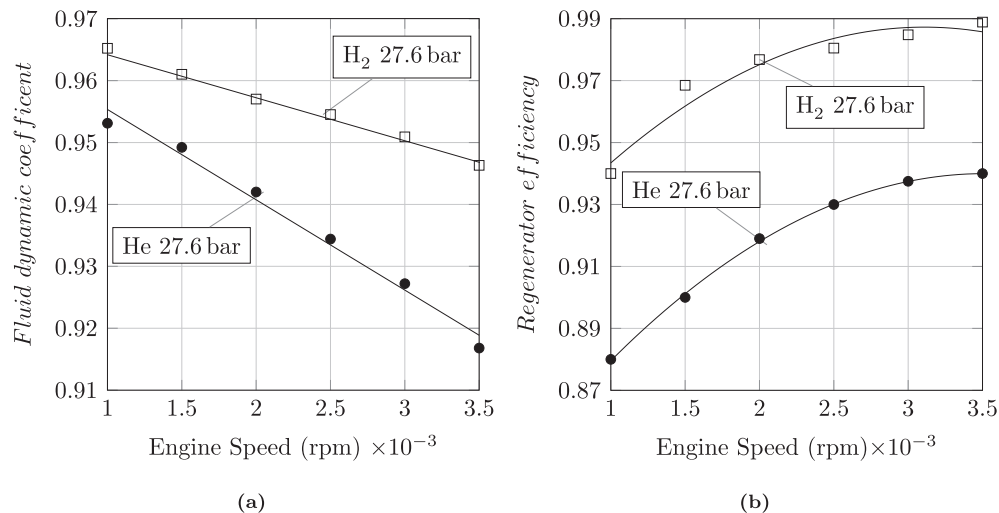


Fig. 4. (a) Optimised Fluid dynamic coefficient η and (b) Regenerator efficiency ε for GPU-3 engine working with helium and hydrogen with fixed mean pressure (27.6 bar).

Table 2

Molecular mass, critical temperature, critical pressure for pure fluids and binary interaction parameter in some CO₂ mixtures.

Fluid	M_M (g/mole)	T_{cr} (°C)	P_{cr} (bar)	k_{ij} CO ₂
CO ₂	44.01	31.06	73.83	–
Xe	131.293	16.59	58.41	–
C ₂ H ₂	26.038	35.15	61.38	–
CH ₄	16.06	–82.58	45.99	0.0919
CHF ₃	70.01	25.86	48.16	0.00824
C ₂ H ₆	30.07	32.17	48.72	0.1322
C ₇ H ₈	92.14	318.6	41.08	0.00824
C ₂ F ₆	138.01	19.88	30.43	0.06814
C ₃ F ₈	188.02	71.90	26.80	–
C ₈ F ₁₈	438.06	229.05	14.78	0.07593
CBrF ₃	148.91	67.00	39.70	–
C ₂ HF ₅	120.02	66.02	36.20	–
SF ₆	146.06	45.54	37.60	–

In the case of the mixtures CO₂–CHF₃, CO₂–C₂F₆ and CO₂–C₈F₁₈, the coefficients k_{ij} were calculated by means of a suitable regression from vapour–liquid balance data taken from Refs. [23–26].

The work in Stirling engines is proportional to the difference between the minimum pressure and the maximum pressure in the cycle. As shown by Fig. 5, for reduced temperatures close to critical and for $P_{r,max} > 2$ there is a greater pressure variation compared to the ideal gas conditions ($T_{r,K} = 2$). This is due to an inferior mean compressibility of the working fluid, which translates into a significant increase in the useful power that can be extracted from the cycle.

In Ref. [15], it has been observed that, when considering an ideal engine, ($\eta = 1$ e $\varepsilon = 1$) the cycle performance varies greatly from ideal gas to real gas. Considering an engine cycle with $\tau = 2$ with a gas at $T_{r,K} = 1.1$, as the reduced pressure maximum varies, there is a fall in efficiency for $P_{r,max} = 2$ whilst for $P_{r,max} > 6$ it tends once more to the value of an ideal gas case. This is due to the losses of a thermodynamic nature in the regenerator.

In Fig. 6 results are given of the entropic analysis for an ideal engine. In the graph, for the case of hydrogen as working fluid, the performance is independent of the operating pressure (Fig. 6b).

In Fig. 6a, on the contrary, there is a dramatic fall in the rate of performance loss in the Regenerator at around $P_{r,max} = 2.3$. The loss tends to fall as the reduced pressure maximum of the cycle increases.

The figure shows that the losses are mainly in the Regenerator. We can observe from Fig. 7, for a fluid with $T_{r,K} = 1.1$, that the flow temperature from the Regenerator to the Heater is lower than in the case with ideal gas. This has a direct influence on the thermal

power in the Heater, directly connected to the difference ($T_H - T_{RH}$). The temperature reduction T_{RH} is due to the difference in specific heat between the hot and cold flows, caused by the cooling of the flow, dropping to temperatures close to the critical.

3.1. Real gas effects in non ideal Stirling engines

The efficiency of the Regenerator directly influences the engine performance. Fig. 8 shows that the Regenerator introduces performance losses that are inferior in the case with hydrogen (ideal gas) than in that with carbon dioxide, for real gas conditions. As far as the effect of the fluid-dynamic losses is concerned, there is a significant increase of $\Delta\eta_{cy}$ in the Expander and Compressor, compared to the ideal case, for $\eta < 1$. As the minimum temperature of the cycle falls, the negative effect introduced by η tends to diminish progressively.

In order to evaluate the power of the engine cycle, we use an adimensional parameter, known as the Beale number, but defined respect the maximum pressure of the cycle (Equation (14)).

$$Be = \frac{\dot{W}}{P_{max} V_{swf}} \quad (14)$$

From calculations with $\eta = 0.98$ and $\varepsilon = 0.95$, in the case with carbon dioxide at $T_{r,K} = 1.1$, a reduction was observed compared to the maximum value for the ideal engine, of about 20% both for the Beale number and for the cycle performance. For hydrogen (ideal gas), there is a reduction of around 60% in both the performance and the Beale number. In the case with ideal gas, most of the losses are concentrated in the mobile volumes of compression and expansion, whilst, in the case of real gas, they are in the Regenerator. The latter is the most critical element in obtaining efficiency in the engine cycle.

It should be observed that the losses in mechanical power due to the pressure losses are distributed, above all, in the heat exchangers and, given its geometric characteristics, mainly in the regenerator. For the sake of simplicity, in our calculation model, the coefficient η takes into account the overall effect of the pressure losses, reducing the work of expansion and increasing that of compression and expansion.

4. Stirling engine working with real gases for low temperature applications

The possibility of working under optimal conditions of temperature and reduced pressure is dependent on the characteristics of the heat sources available and the cooling system.

Thus, there is no optimal fluid for all applications, instead, it must be chosen on the basis of the various working conditions, and compatibly with the thermal and chemical stability of the fluid itself.

Below, engine performances are evaluated with two values of maximum temperature, $T_{max} = 300^\circ\text{C}$ and $T_{max} = 150^\circ\text{C}$, two values of temperature for the cold well, one of reference for applications that just produce mechanical power ($T_{min} = 40^\circ\text{C}$) and the other for co-generative applications ($T_{min} = 80^\circ\text{C}$).

Finally, three values were assumed for the maximum pressure of the fluid inside the engine (100 bar, 150 bar, 200 bar). In general, the maximum pressure has a positive effect on the engine performance, both in terms of specific power and efficiency. Increasing the maximum pressure, though, means increasing the costs of building the engine, since components must be designed to withstand greater mechanical stress.

Lastly, the analysis was made by fixing $\varepsilon = 0.95$ and $\eta = 0.98$. These parameters must be chosen in accordance to the global

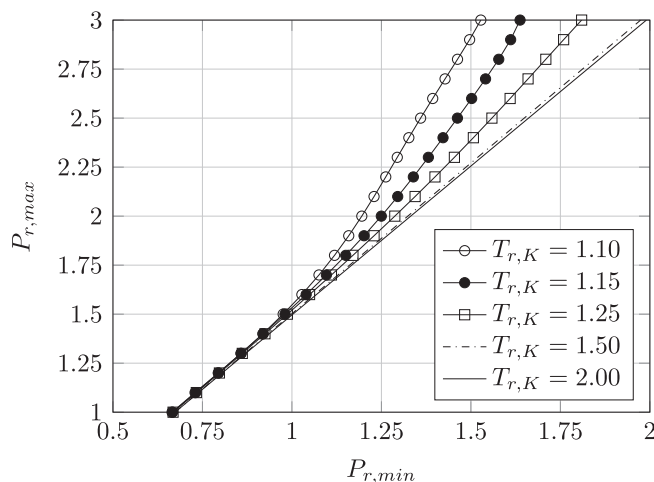


Fig. 5. Xenon minimum reduced pressure $P_{r,min}$ as function of $P_{r,max}$ and $T_{r,K}$ ($\eta = 1$, $\varepsilon = 1$).

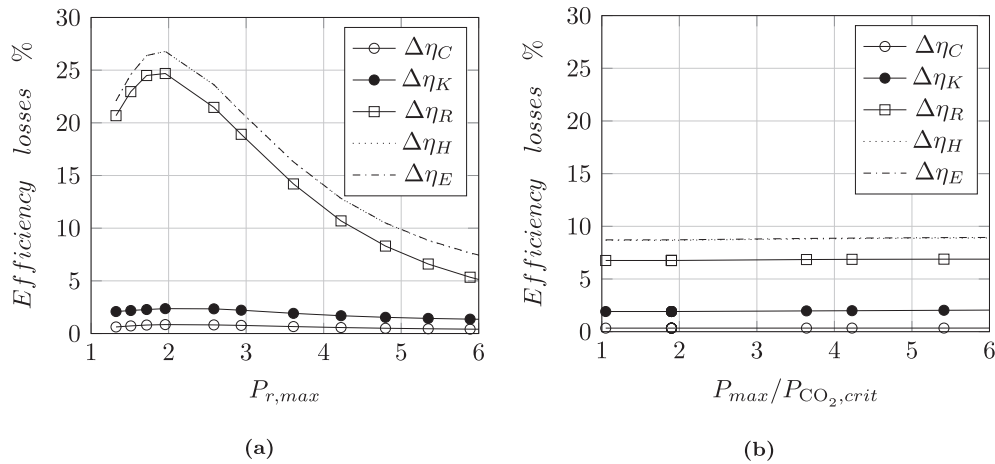


Fig. 6. Efficiency losses with adiabatic model of an engine working with (a) real gas effect (CO₂ $T_{r,K} = 1.1$) and (b) with ideal gas (H₂) ($\eta = 1$; $\epsilon = 1$; $\tau = 2$).

performance (power and efficiency) of existing engines and so experimental data are necessary to find their right values. On the other hand, because there are no experimental data of Stirling engines working with a fluid under real gas behaviour, the estimated and selected values of these two parameters are only indicative and we assumed they are valid for both real and ideal gas Stirling engine.

Results obtained with the different working fluids are to be found in Table 3. Note how the choice of the working fluid affects both the useful power and the engine performance.

For example, the carbon dioxide performs better in terms of useful power, but the engine that it pressurises has a significantly poorer performance than those using other fluids. Under the working conditions considered, the thermodynamic state of carbon dioxide is close to the zone of greatest inefficiency in the regenerator, thereby requiring a significant amount of heat in order to reach the maximum temperature of the engine cycle (Fig. 7a).

Reducing the pressure from 200 bar to 150 bar (Table 3), we can see that, for the ideal gas, the useful power that can be extracted drops by about 20%, without significant variation in terms of efficiency.

As far as real gases are concerned, the pressure reduction means a variation of the working $P_{r,max}$, which causes the work point of the fluid to move on the thermodynamic plane, receding from (or approaching) the point of optimum work with respect to the

previous case. Table 3 reveals, for example, that at 200 bar the engine pressurised by xenon has both a superior performance and mechanical power to that of C₂F₆, but the situation is reversed at 150 bar.

The effect of the different reduced conditions of work can be seen even more clearly when observing the behaviour of C₂F₆ which, with these temperature values and pressure equal to 100 bar, offers even better performances than ethane, which, in the previous cases, has higher values in terms of both performance and mechanical power.

We now consider the case of relatively low maximum temperature in the cycle (about 150 °C). Heat sources with these characteristics are very common in industrial applications, whether for production necessity or as waste energy from production processes [27]. This segment of the market is of great interest from the point of view of wasted energy, what is more, these temperature values can be obtained with solar energy plants at low level concentration. Compared to the previous cases, temperature variation between maximum and minimum in the cycle is lower, which leads to a significant reduction in the useful work obtainable from the engine cycle. Reducing the maximum temperature of the cycle gives smaller pressure variations and, consequently, reduces the mechanical power that can be extracted from the cycle. As confirmation of this, it can be seen that reducing the maximum temperature of the engine cycle, to guarantee reaching the maximum pressure, it is necessary to insert a greater fluid mass than in high

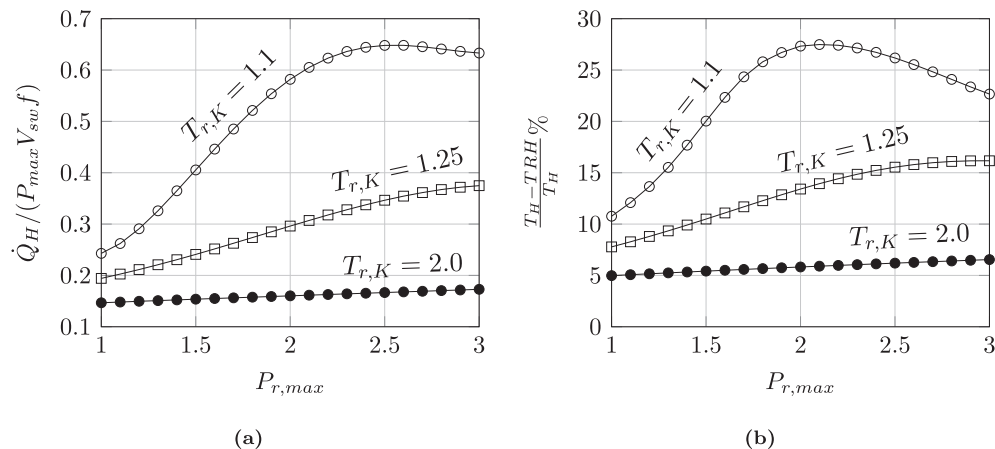


Fig. 7. (a) Non dimensional Heater thermal power and (b) Regenerator–Heater temperature interface ratio of Stirling engines working with xenon ($\eta = 1$; $\epsilon = 1$).

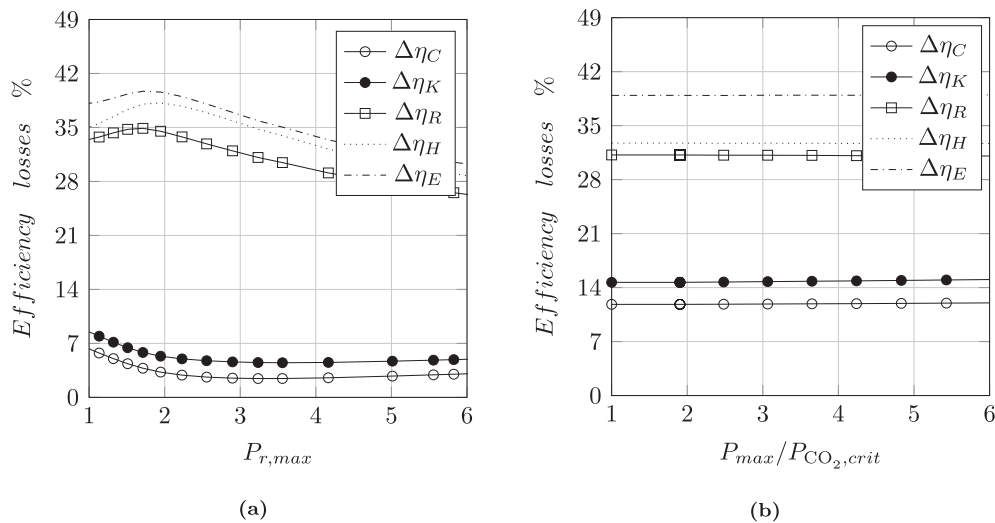


Fig. 8. Efficiency losses with adiabatic model of an engine working with (a) real gas effect (CO_2 $T_{r,K} = 1.1$) and (b) with ideal gas (H_2) ($\eta = 0.98$; $\varepsilon = 0.05$; $\tau = 2$).

temperature cycles (Table 4). Table 4 shows also that, fixing the maximum pressure of the cycle, fluids with greater molar mass imply a proportional increase of the fluid mass charged in the engine. This could be a negative factor when opting to use costly working fluids. This is another reason that make mixture with an high concentration of CO_2 (cheaper than helium, hydrogen) interesting as working fluid for Stirling engine. Moreover, Stirling engine pressurised with hydrogen or helium have severe leakage problems and they must be cyclically pressurised. This problem could be more easily kept under control using heavier fluids as working fluid.

In Table 5 we report the results obtained with a maximum temperature of 150°C . Under these working conditions, the real gases can offer good performances in both terms of useful power and conversion efficiency. In fact, while the ideal gas engine does not give useful power, the real gas cycles, which work across the critical point, offer discreet performances.

When analysing Stirling engines for co-generative applications, it is useful to consider other working fluids than those above, with higher critical temperature values, in order to operate always within the optimum work zone.

Table 6 reports the performance values for the fluids analysed for the co-generative processes, with a heat source temperature of 300°C .

We observed that CBrF_3 offers the best performance, compared to other fluids, for maximum pressures of 200 bar and 150 bar whilst, when the maximum pressure drops to 100 bar, the best-performing fluid is C_3F_8 . This is because the optimal thermodynamic conditions depend on both the reduced temperature and the

reduced pressure. In fact, over a certain value of the latter, there is a progressive fall-off in the engine performance caused by the increased heat needed to heat up the greater mass of fluid inserted, which is not counterbalanced by a proportional increase in useful power.

Fig. 9 shows the diagrams P – V , which have been adimensioned with respect to the maximum values of pressure and volume, for the volumes of compression and expansion. These graphs indicate the areas of the diagram in which the engine produces work (the darker ones, indicated in the legend by W_e) and those where the engine requires pumping work (the lighter ones, indicated in the legend by W_c). The difference between these areas represents the useful work produced by the engine. Comparing these graphs, it can be seen how using the fluids in optimal thermodynamic conditions causes the diagrams to change considerably.

5. Carbon dioxide mixtures as working fluids

The use of mixtures as working fluids is particularly interesting as it makes the most of the effects of real gas, because a variation in the mixture composition is directly correlated to a variation in the critical point [28]. This characteristic means that working fluids can be chosen for their suitability for each individual application. Our analysis shows that mixtures of carbon dioxide and organic fluids give especially interesting results.

There are, essentially, two major limits to using carbon dioxide as the working fluid in these engines: a high critical pressure (73.83 bar) and a relatively low critical temperature (31.06°C). On the other hand, though, it is generally considered to be a stable fluid in the presence of common materials, even at high temperatures

Table 3

Cycle performance with different working fluids ($\eta = 0.98$, $\varepsilon = 0.95$) with $T_{\text{max}} = 300^\circ\text{C}$.

Fluid	$T_{r,K}$	P_{cr} (bar)	P_{max}					
			200 bar		150 bar		100 bar	
			$\dot{W}$ (kW)	$\eta_{cy}\%$	$\dot{W}$ (kW)	$\eta_{cy}\%$	$\dot{W}$ (kW)	$\eta_{cy}\%$
H_2	9.44	12.8	0.586	8.68	0.43	8.7	0.3	8.68
CO_2	1.03	73.83	6.53	12.8	3.63	9.13	$P < P_{\text{sat}}$	$P < P_{\text{sat}}$
C_2H_6	1.026	48.72	5.83	18.7	4.91	15.5	3.22	11.2
Xe	1.08	58.41	5.13	17.9	3.32	13.2	1.24	9.10
CHF_3	1.05	48.16	5.93	16.9	4.54	13.5	2.49	9.65
C_2F_6	1.07	30.43	4.50	15.0	3.72	13.8	2.91	12.2

Table 4

Working fluids mass ($\eta = 0.98$, $\varepsilon = 0.95$) with $P_{\text{max}} = 200$ bar.

Fluid	$M_{300^\circ\text{C}}$ (g)	$M_{150^\circ\text{C}}$ (g)	$\Delta M/M_{150^\circ\text{C}}\%$
H_2	2.238	2.567	12.79
CO_2	75.77	102.5	26.10
C_2H_6	47.78	66.2	27.82
Xe	216.3	289.7	25.33
CHF_3	105.3	142.0	25.80
C_2F_6	173.4	226.4	23.40

Table 5

Cycle performance with different working fluids ($\eta = 0.98$, $\varepsilon = 0.95$) with $T_{\max} = 150$ °C.

Fluid	$P_{\max}$ (bar)					
	200		150		100	
	$\dot{W}$ (kW)	η_{cy}	$\dot{W}$ (kW)	$\eta_{cy}\%$	$\dot{W}$ (kW)	$\eta_{cy}\%$
CO ₂	3.44	11.7	2.15	6.60	$P < P_{\text{sat}}$	$P < P_{\text{sat}}$
C ₂ H ₆	2.48	12.8	2.12	12.3	1.77	9.15
Xe	2.08	11.0	1.37	7.50	0.46	3.44
CHF ₃	2.64	12.1	2.20	10.7	1.46	7.15
C ₂ F ₆	1.89	10.0	1.68	10.0	1.38	9.47

Table 6

CHP cycle performance with different working fluids ($\eta = 0.98$, $\varepsilon = 0.95$) with $T_{\max} = 300$ °C.

Fluid	$T_{r,K}$	P_{cr} (bar)	$P_{\max}$					
			200 bar		150 bar		100 bar	
			$\dot{W}$ (kW)	$\eta_{cy}\%$	$\dot{W}$ (kW)	$\eta_{cy}\%$	$\dot{W}$ (kW)	$\eta_{cy}\%$
CO ₂	1.16	73.83	2.24	9.74	1.19	8.21	0.46	7.12
Xe	1.218	58.41	2.25	12.11	1.24	9.39	0.51	8.05
CBrF ₃	1.038	39.70	3.94	15.90	3.45	14.79	2.53	11.08
C ₃ F ₈	1.023	26.8	3.49	12.70	3.44	12.10	2.55	11.13
C ₂ HF ₅	1.041	36.20	3.97	14.15	2.94	13.18	2.61	10.67
F ₆ S	1.108	37.60	3.32	12.63	2.64	11.41	1.52	8.75

and pressures, as well as easily available in nature and non-inflammable.

In the previous paragraphs, entropic analysis and comparison with various working fluids has shown that the condition for optimum performance and maximum specific power in cycles with working fluids that behave like real gas is obtainable with values of $T_{r,K}$ close to the critical temperature and for maximum pressure values greater than 4–5 times the critical pressure of the working fluid.

Consequently, to obtain the same efficiency as other working fluids, it would be necessary to pressurise an engine using CO₂ as its working fluid to over 300 bar. By properly mixing carbon dioxide with other substances, it is possible to get high efficiencies even at lower working pressures, while, at the same time, guaranteeing high specific power values.

On the other hand, owing to the possible decomposition of organic fluid in high temperature applications, the use of CO₂ and organic fluid mixtures as working fluid is practicable only with heat sources at maximum temperature below 400 °C [16]. Moreover, with these fluids, the preliminary engine design can't benefit on

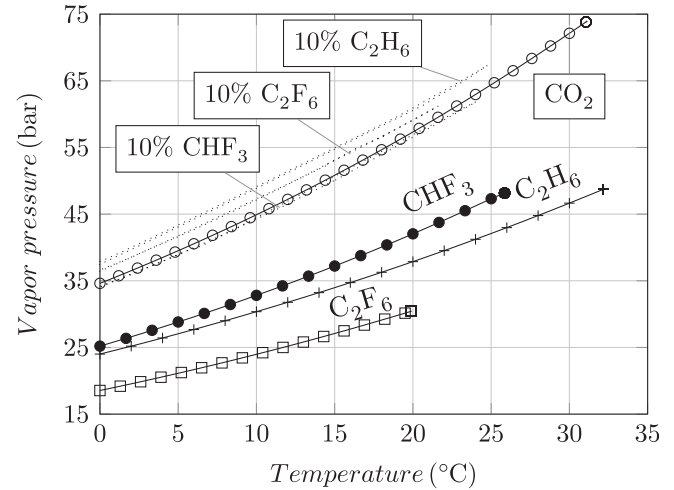


Fig. 10. Vapour pressure diagram of a CO₂, some organic fluids and their mixtures with CO₂.

preliminary design methods based on both data or simulations available for pure fluid in ideal gas conditions.

5.1. Stirling engine with mixtures as working fluids

Choosing the most suitable fluid mixture for the various applications depends on evaluating the dew and bubble curves.

Fig. 10 represents several dew and bubble curves for various mixtures with carbon dioxide. The characteristics of these fluids means that the temperature glide is very small. Considering, for example, a mixture made up for 90% by Ref. CO₂ and for 10% by ethane, at a pressure of 55 bar, we get a temperature glide, calculated by means of the Peng–Robinson equation using the coefficients of binary interaction reported in Table 2, equal to 0.406 °C.

The mixtures considered have a lower cricondenbar than the P_{cr} of the CO₂ and a slightly lower cricondetherm than the T_{cr} of the CO₂: from 73 bar to about 60 bar (a drop of 18%) and from 304 K to 296 K (a drop of 3%), respectively.

As a result, by using certain mixtures, with the proper concentrations of organic fluid, it is always possible to keep the maximum operating reduced pressures around the optimal zone.

Fig. 11 reports some of the results obtained with a variety of fluid mixtures and with various concentrations. Performances are

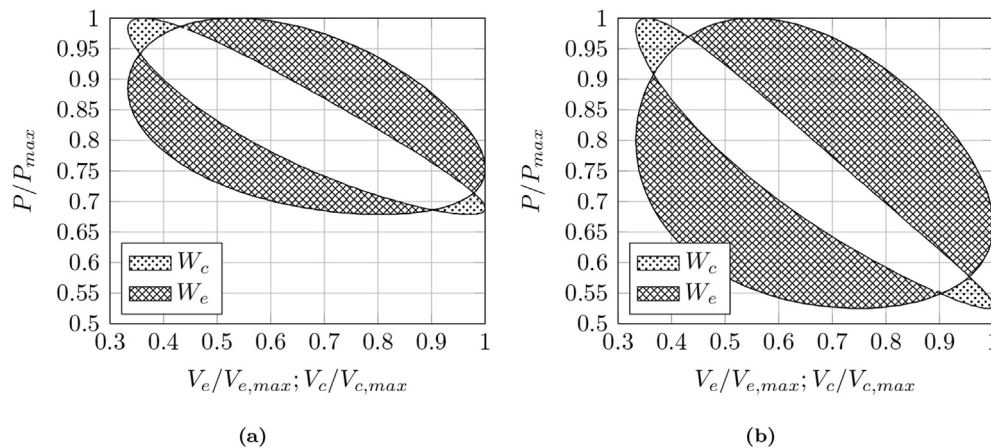


Fig. 9. Expansion and compression cycles in the P – V plane with $\eta = 0.98$, $\varepsilon = 0.95$, $P_{\max} = 150$ bar and $T_K = 80$ °C for (a) CO₂ ($T_{r,K} = 1.16$, $P_{r,\max} = 2.04$), (b) CBrF₃ ($T_{r,K} = 1.04$, $P_{r,\max} = 3.76$).

reported in terms of mechanical power and efficiency of the various cycles, compared to those obtained with pure carbon dioxide, for a maximum temperature of 300 °C and a minimum of 40 °C and maximum pressures of 150 bar and 200 bar.

The abscissa axis reports the fluids mixed with carbon dioxide, where the molar fraction is indicated by the number following the fluid chemical formula. In choosing the optimal fluid, the concentration of the organic fluid should be kept as low as possible, for reasons of both cost and environmental impact.

In fact, although these fluids are not particularly reactive or unstable, they have a high GWP (Global Warming Potential) [29] and are harmful to the ozone layer [30].

In the upper part of the Fig. 11a we can analyse the power and efficiency results for an engine pressurised at 200 bar. At maximum pressure, the mixtures with organic fluid concentrations greater

than 10% molar reveal a power extractable from the engine cycle inferior to pure CO₂. Only by reducing concentration levels below 10% is it possible to get comparable power values. On the other hand, by raising the organic fluid concentration, performance improves by around 15 percent, compared to pure CO₂, for molar concentrations of fluid of 25%. However, by reducing the maximum pressure to 150 bar, the diagrams change significantly. In fact, we note that, while in Fig. 11a there is a significant gap between the power calculated with pure CO₂ and the various mixtures, in Fig. 11b there is less variability and, what is more, with the mixture composed by 75%CO₂–25%CHF₃ we have a useful power around 1.15 times greater than pure CO₂. As far as the cycle performances are concerned, we can see the qualitative trend of the diagrams does not change: the mixtures with greater concentrations of fluid additive offer greater efficiency, which tends to diminish as the concentration levels of CO₂ rise.

5.2. Stirling engine for CHP applications with mixtures of fluids

We now turn to the case of the co-generative engine. Analysing the performance of mixtures of carbon dioxide, where the latter is the dominant component, it is necessary to find fluids, which, by their characteristics, profoundly affect the dew and bubble curves, despite being mixed in relatively low concentrations. In fact, the CO₂ has a critical temperature close to 30 °C, so, if we want to bring the limit curve nearer to operating conditions, we need to take into consideration fluids other than those used in the comparison between pure fluids. A first analysis reveals that the most interesting fluids are toluene (C₇H₈) and perfluorooctane (C₈F₁₈). Observing the state diagrams reported in Fig. 12, we note that the effect of the various mixed fluids is significantly different. The graph reports the curves representing the liquid and saturated steam pressures for three pure fluids, for a compound mixture 80%CO₂–20%C₈F₁₈ and another compound of 95%CO₂–5%C₇H₈. It can be seen how mixing just a small percentage of toluene with carbon dioxide is sufficient to bring about a major change in the state diagram for the main fluid. To get similar dramatic changes with the perfluorooctane, it is necessary to mix in a greater quantity of fluid additive. As confirmation of this, it has been verified that with percentages of toluene greater than 6%, we obtain a fluid with a vapour pressure diagram that would make it unusable since the mixture changes to a two-phase state.

To conclude our analysis of the co-generative cycle, Fig. 13 presents the results for engines with a maximum pressure of 200 bar and 150 bar. The graphs also reveal that while the mixtures of CO₂ and perfluorooctane produce significant effects on the

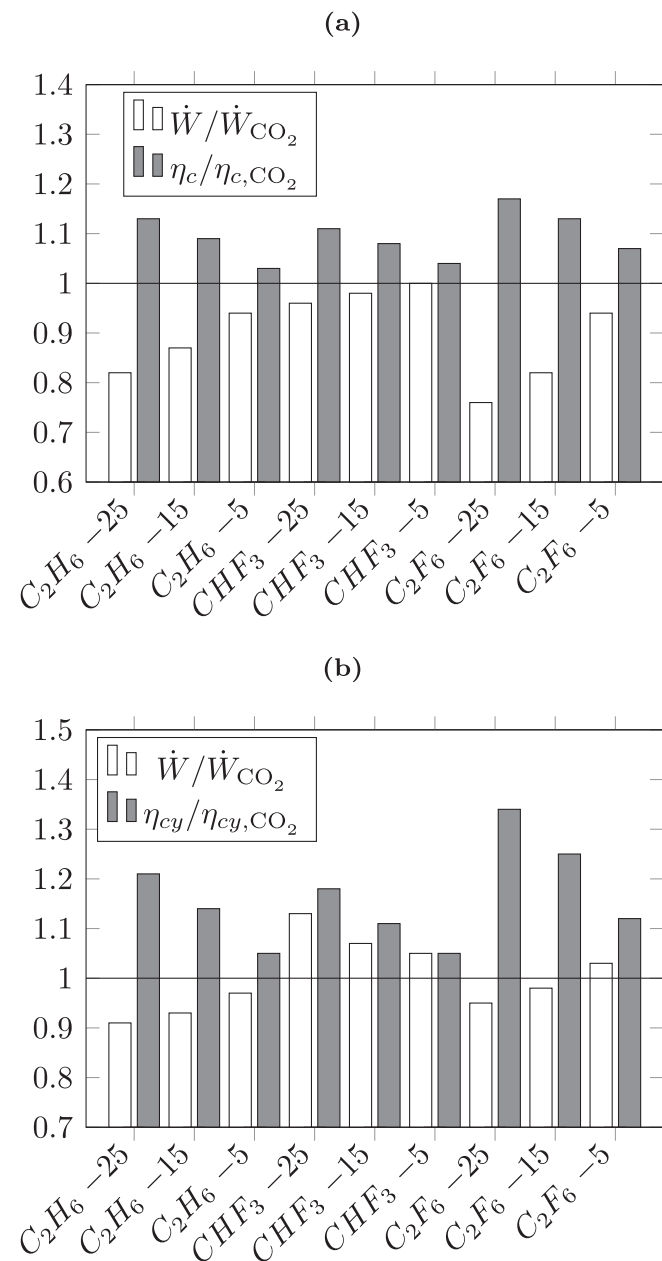


Fig. 11. Power production and efficiency for engines working with $T_{\text{max}} = 300$ °C, $T_{\text{min}} = 40$ °C, $\eta = 0.98$, $\epsilon = 0.95$ and different carbon dioxide mixtures compared with engine working with carbon dioxide (a) $P_{\text{max}} = 200$ bar and (b) $P_{\text{max}} = 150$ bar.

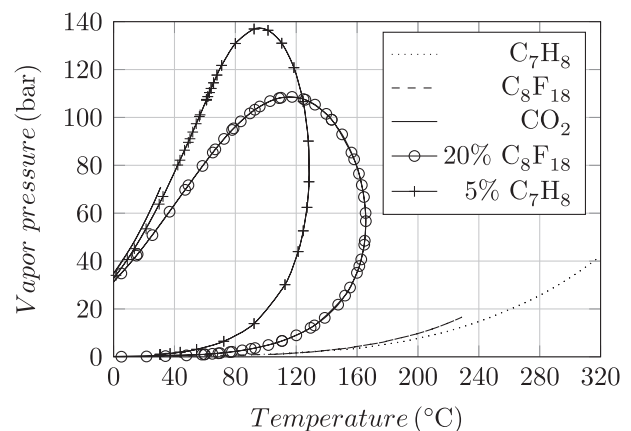


Fig. 12. Vapour pressure diagram of carbon dioxide, toluene, perfluorooctane and a mixture of 95%CO₂–5%C₇H₈ and mixture of 80%CO₂–20%C₈F₁₈.

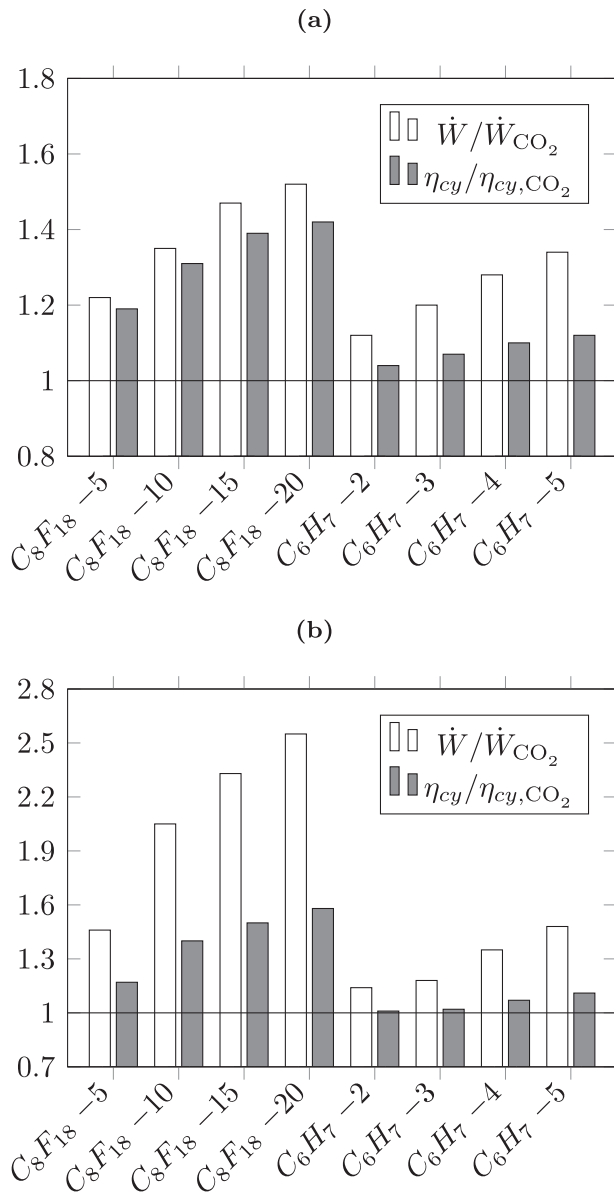


Fig. 13. A Power production and efficiency for engines with $T_{\max} = 300$ °C, $T_{\min} = 80$ °C, $\eta = 0.98$, $\epsilon = 0.95$ and working with different carbon dioxide mixtures compared with engine working with carbon dioxide (a) $P_{\max} = 200$ bar and (b) $P_{\max} = 150$ bar.

useful power and engine efficiency, mixtures with toluene do not offer similar interest. The graphs also highlight a notable increase in both mechanical power and efficiency of the mixtures compared to the pure CO_2 . In fact, the carbon dioxide barely responds to the effects of real gas, making it of very limited interest for this application. By contrast, adding the mixture of C_8F_{18} , we get performances comparable with the pure fluids, which, in the first part of our analysis, were optimal for these working conditions. So, using C_3F_8 as the working fluid, we obtain a useful power of around 2.55 kW and an efficiency of 11.4%, whilst with the optimal compound mixture with 20% perfluorooctane, the useful power is about 2.39 kW and efficiency stands at 11.8%.

6. Conclusion

The benefits derived from using a working fluid under real gas conditions are obtained for T_K close to the critical temperature of

the fluid itself and for maximum pressures of the cycle 3 to 4 times greater than the critical pressure of the working fluid. Analysis carried out during our research has shown that using fluids in real gas conditions in Stirling engines prompts a significant increase in the specific power of the engine, 2–3 times greater than that of ideal gas. This effect is due to the greater pressure variation that occurs with a fluid under real gas conditions, compared to ideal gas conditions, during the phases of the engine cycle.

The effects of real gas cause a sharp drop in efficiency inside the regenerator due to the heat exchange between the real gas flows. These losses, though, tend to diminish as the reduced pressure maximum of the cycle increases. The effect of this loss of useful power and efficiency of the regenerator varies greatly in its impact according to the thermodynamic state of the working fluid. This analysis has shown that the optimal fluid for an engine cycle depends on the characteristics of the heat source and the maximum pressure at which the engine is designed to operate. In general, carbon dioxide has given very interesting results. As well as being a fluid with excellent characteristics of thermal stability, easily available and non-toxic, it offers high performance levels in terms of specific power. Unfortunately, this fluid has a very high critical pressure and, given that the regenerator inefficiencies can only be exceeded for pressure $>(3/4)P_{cr}$, this means it would be necessary to pressurise the engine to over 280 bar. Exploiting mixtures of CO_2 and organic fluids, though, enables us to avoid this drawback. In fact, the mixture of two fluids generates new compounds, which, if mixed together in the proper concentrations, can provide the optimal thermodynamic conditions for the various working conditions in the Stirling engines considered in this research. Furthermore, mixing CO_2 with the right fluids means the critical temperature of the mixture can be raised, which also makes the use of carbon dioxide interesting for co-generative engines, where, given its low critical temperature, it had not been considered an appropriate fluid for exploiting the effects of real gas.

An experimental analysis on Stirling engine working with real gas effect is strongly recommended for future work over this theme.

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